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SUMMARY

An investigation was conducted to determine the effect of carburetor dry-air temperature on the cooling characteristics of a typical full-scale air-cooled engine cylinder mounted on a single-cylinder crankcase. The variable carburetor-air temperature tests, conducted at constant charge-air weight flow and constant cooling-air pressure drop, covered carburetor-air temperatures from 100° to 310° F at fuel-air ratios of 0.08 and 0.10. The NACA cooling correlation method was used as the basis for analysis. The effect of carburetor-air temperature on the cooling characteristics is shown by the variation of mean effective gas temperature with carburetor dry-air temperature.

It was found that for the fuel-air ratios investigated the mean effective gas temperature varied about 1/2° F for the cylinder head and about 1/4° F for the cylinder barrel with each 1° F variation in carburetor dry-air temperature. The results obtained are believed to indicate a trend but the exact values are probably not applicable to other than the cylinder used for these tests.

The observed variation in cylinder cooling due to the carburetor dry-air temperature was applied to the full-scale engine, and curves are presented for determining the mean effective gas temperature of the cylinder head for given values of fuel-air ratio, engine speed, and carburetor inlet-air temperature. Curves are also presented to show the change in head temperature resulting from a change in mean effective gas temperature.

INTRODUCTION

The air cooling of a finned cylinder consists in a balance of two processes: (1) the transfer of heat from the combustion gases to the external cylinder wall and (2) the transfer of heat from the finned surfaces to the cooling air. The internal and the external cooling processes are analyzed in reference 1 and a method is developed for relating the average cylinder temperatures to the engine variables and the cooling-air conditions.

The instantaneous combustion-gas temperature, upon which the internal heat-transfer process depends, varies throughout the engine cycle and because of the complexity of the variation a mean "effective" gas temperature was used instead of the true mean combustion-gas temperature to establish the general cooling relation of reference 1. The mean effective gas temperature for a given engine has been shown to depend primarily on the fuel-air ratio; to depend somewhat less on the carburetor-air temperature, the exhaust manifold pressure, and the spark timing; and to be independent of the engine speed. Although the absolute value of the mean effective gas temperature is difficult to determine with any degree of accuracy, the variations from a given value caused by the fuel-air ratio, carburetor-air temperature, exhaust-manifold pressure, and spark-timing may be determined quite accurately and are of great importance in proper interpretation of cooling data.

The variation of the mean effective gas temperature with fuel-air ratio has been adopted as a portion of the standard cooling test and has been thoroughly investigated for both single-cylinder and multicylinder engines. The effect of spark timing (references 1 and 2) appears to be small within the operating range of spark settings. Reference 2 also shows the effect of exhaust-manifold pressure to be slight. Available data, although meager, indicate that the mean effective gas temperature appreciably increases with an increase in carburetor-air temperature (references 1, 2, and 3), but the rate of increase appears to vary between cylinders of different design. At present the practice is to assume that the mean effective gas temperature for the cylinder head varies 0.8° F for each 1° F change in carburetor-air temperature for all cylinders.

The present investigation was conducted at the Langley Field laboratory of the NACA on a typical air-cooled cylinder mounted on a single-cylinder crankcase to determine further the effect of carburetor-air temperature on mean effective gas temperature. The test results are presented in a form convenient for use in multicylinder-engine work.

APPARATUS

Test equipment. - A front-row cylinder from a widely used conventional air-cooled 18-cylinder radial engine was mounted on a single-cylinder crankcase as shown in figure 1. The cylinder compression ratio was 6.7, the spark setting was 20° B.T.C., and the valve timing was as follows:

Valve	Opens	Closes
Intake	9° B.T.C.	62° A.B.C.
Exhaust	80° B.B.C.	18° A.T.C.

An electric dynamometer and a water brake were used for power absorption.

Cooling air was supplied to the cylinder by a centrifugal blower. The cylinder baffles used were not standard but were designed to maintain a constant free-flow area along each of the interfin air passages. (See reference 4.) The baffle changes should have a negligible effect on the test results because a change in external cooling should not substantially affect the internal-cooling processes.

Charge air was provided at the desired manifold pressure by an auxiliary blower; for the tests at varying carburetor-air temperature the charge air was maintained within ±2° F of the desired value by an automatically controlled electric heater. The carburetor consisted of an injection nozzle located in a venturi section and was connected to the cylinder-intake port by a short converging elbow.

Measuring devices. - The charge-air weight flow was measured by a thin-plate orifice located in a straight section of pipe upstream of the heater. The pressure drop across the orifice was indicated by a water manometer and an iron-constantan thermocouple was used to measure the air temperature at the orifice. The carburetor-air temperature was obtained by means of an unshielded iron-constantan thermocouple located in the center of the carburetor-air duct above the carburetor. An automatic weighing stand measured the fuel flow and the fuel-air ratio was calculated from the measured fuel and charge-air weight flows. Engine speed and torque were obtained with standard test equipment.

The cylinder temperatures were measured by iron-constantan thermocouples peened into the outer wall at the positions shown in figure 2. The temperature of the rear spark-plug gasket was measured by a gasket-type thermocouple. All iron-constantan thermocouples were used in conjunction with a potentiometer and the cold junctions were located in an insulated box, the temperature of which was measured by an alcohol-in-glass thermometer.

The cooling-air pressure drop, including the baffle-exit loss, was measured as the pressure difference between the atmosphere and a static-pressure ring located in the large duct ahead of the cylinder. The cooling-air temperature was measured by thermocouples located upstream and downstream of the cylinder.

METHODS AND TESTS

In order to determine the effect of a particular variable on the mean effective gas temperature, it was first necessary to establish by tests the cooling relation for the test cylinder.

Cooling tests. - The procedure followed for establishing the relation of the cylinder temperature to the cooling air and the engine operating conditions was the same as that outlined in reference 5. The cooling-air flow and the charge-air weight flow were separately varied for the tests at a constant fuel-air ratio. The test conditions were as follows:

Engine speed (rpm)	Charge-air weight flow (lb/min)	Fuel-air ratio	Carburetor dry-air temperature (°F)	Cooling-air pressure drop (in. water)
2100	6.25	0.08	80-96	2.5-27.5
2100	4.0-10.8	.08	89-93	15

The cooling relations (fig. 3) derived from the present data are for the head and barrel, respectively:

$$\frac{T_h - T_a}{T_g - T_h} = 1.727 \frac{W_c^{0.535}}{(\sigma_{av} \Delta p)^{0.32}} \tag{1}$$

$$\frac{T_b - T_a}{T_g - T_b} = 2.669 \frac{W_c^{0.637}}{(\sigma_{av} \Delta p)^{0.40}} \tag{2}$$

where

T_h average cylinder-head temperature, $^{\circ}F$

T_b average cylinder-barrel temperature, $^{\circ}F$

T_a cooling-air temperature upstream of cylinder, $^{\circ}F$

T_g mean effective gas temperature, $^{\circ}F$

W_c charge-air weight flow, pounds per second

σ_{av} ratio of average cooling-air density to NACA standard sea-level density, ρ/ρ_0

Δp cooling-air pressure drop across cylinder, inches of water

Excellent consistency between the test data and equations (1) and (2) are shown in figure 3. These equations vary somewhat from those determined in previous investigations with similar cylinders. (See references 5 and 6.) This variation can be attributed in part to the differences in baffle arrangements and instrumentation.

The effect of fuel-air ratio on the mean effective gas temperature was determined at the following conditions:

Engine speed (rpm)	Charge-air weight flow (lb/min)	Fuel-air ratio	Carburetor dry-air temperature ($^{\circ}F$)	Cooling-air pressure drop (in. water)
2100	Varied	0.062-0.105	100	15
2100	6.28	.062- .105	100	15
2100	8.73	.063- .097	100	15

The values of T_g for the cylinder head and barrel were calculated from equations (1) and (2) and the variations with fuel-air ratio are shown in figure 4 for a carburetor dry-air temperature of 100° F.

Carburetor-air-temperature tests. - The effect of carburetor-air temperature on mean effective gas temperature was determined from tests conducted at the following conditions:

Engine speed (rpm)	Charge-air weight flow (lb/min)	Fuel-air ratio	Carburetor dry-air temperature (°F)	Cooling-air pressure drop (in. water)
2100	6.4	0.080	102-309	15
2100	6.7	.100	102-311	15

The charge-air weight flow and the cooling-air pressure drop were held constant in order that the effects of variables other than carburetor-air temperature would be eliminated. The exhaust-manifold pressure was constant for all tests. The carburetor-air-temperature tests were run at two fuel-air ratios to determine whether a change in fuel-air ratio would affect the variation of mean effective gas temperature with carburetor dry-air temperature. The values of T_g were calculated from equations (1) and (2).

RESULTS AND DISCUSSION

Effect of carburetor-air temperature on mean effective gas temperature. - The variation of mean effective gas temperature T_g with carburetor dry-air temperature T_c is shown in figure 5 for the head and the barrel at fuel-air ratios of 0.08 and 0.10. The values of T_g vary linearly with T_c and the slopes of the curves, 0.56 for the head and 0.24 for the barrel, are the same for both fuel-air ratios. The variation of T_g with T_c is apparently independent of the fuel-air ratio in the rich range. For this cylinder a rise of 1° F in the carburetor dry-air temperature results in a rise of 0.56° F in the head mean effective gas temperature and 0.24° F in the barrel mean effective gas temperature.

The results of references 1, 2, and 3 indicate that the relation between T_g and T_c varies for different cylinder designs. The relation between T_g and T_c derived in this report therefore applies only to the cylinder design tested.

Effect of basing the cooling equation on a single critical temperature. - The foregoing variation of T_g for the cylinder head with T_c was obtained from the cooling equation based on the average head temperature. Cooling correlations for multicylinder engines, however, are often based on the use of a single critical temperature measurement on each head, which may be either more or less sensitive to a particular variable than the average head temperature. It is therefore necessary to determine the conditions under which the relation between the mean effective gas temperature and the carburetor dry-air temperature is independent of the single critical temperature used as the basis of the correlation equation and to show further that these requirements are satisfied by any of the external cylinder-head temperatures for the conditions of the present tests.

These conditions can be determined in the following manner: For any external head temperature T_x the linear relation

$$(T_x - T_a) = a + b\,(T_h - T_a)$$

is generally true. The foregoing equation can be rewritten as

$$1 - \frac{a}{(T_x - T_a)} = b\left(\frac{T_h - T_a}{T_x - T_a}\right) \tag{3}$$

where a and b are constants.

By use of equations (1) and (3) it may be shown that the ratio R of the changes in mean effective gas temperature with carburetor-air temperature based on the individual external head temperature and on the average head temperature is given by

$$\frac{\partial T_{g_x}/\partial T_c}{\partial T_g/\partial T_c} \equiv R = b\left(\frac{T_h - T_a}{T_x - T_a}\right)\left(\frac{T_{g_x} - T_a}{T_g - T_a}\right) \tag{4}$$

where T_{g_x} is the mean effective gas temperature based on T_x in °F.

Because the ratio $\dfrac{T_{g_x} - T_a}{T_g - T_a}$ will always be very nearly equal to unity, equation (4) can be closely approximated by

$$R = b\left(\frac{T_h - T_a}{T_x - T_a}\right) \tag{5}$$

From equations (3) and (5) it follows that, when the intercept a equals zero, the ratio R equals unity. Under this condition the relation between the mean effective gas temperature and the carburetor dry-air temperature determined by using the average head temperature will apply to a correlation based on a single critical temperature.

For all the operating variables covered by the present tests, the temperature of the rear spark-plug gasket satisfied equation (3) and the intercept a was zero. (See fig. 6.) It is possible, however, that the intercept a may not be zero for variables other than those investigated in these tests. In order to substantiate the foregoing analysis, the cooling-correlation equation based on the temperature of the rear spark-plug gasket was established, and the variation of mean effective gas temperature with carburetor dry-air temperature was investigated. The results (fig. 7) exhibit the same relation that was previously obtained using the average head temperature; thus the analysis is shown to be valid.

A number of external cylinder-head temperatures were investigated and were found to satisfy equation (3). The values of the intercepts were such that the maximum variation of the slope of the curves of T_g against carburetor-air temperature from the value given by figure 5 was ±0.05. It may be concluded therefore that with sufficient accuracy the relation between T_g and T_c is independent of the external cylinder-head temperature used as the basis for the correlation equation.

APPLICATION OF RESULTS

The results of the present tests are of practical interest only in their application to multicylinder engines. Because the true dry inlet-manifold temperature is difficult to measure in a multicylinder engine and is indefinite because of the evaporation of an unknown amount of fuel, the effective dry manifold temperature t_m was adopted as the criterion of the cylinder inlet-air temperature. (See reference 5.) The effective dry manifold temperature, which corresponds to the carburetor dry-air temperature measured in the single-cylinder tests, is defined as the sum of the carburetor inlet-air temperature and the temperature rise through the engine supercharger, neglecting the effect of fuel vaporization. The effective dry manifold temperature (reference 5) may be expressed as

$$t_m = t_c + \frac{U^2}{gc_pJ} \tag{6}$$

where

t_m effective dry manifold temperature, °F

t_c carburetor inlet-air temperature, °F

U impeller tip speed, feet per second

g acceleration of gravity, feet per second per second (32.2)

c_p specific heat of air at constant pressure, Btu per pound per °F (0.24)

J mechanical equivalent of heat, foot-pounds per Btu (778)

The term $\frac{U^2}{g c_p J}$ is a close approximation to the theoretically correct expression for the dry-air temperature rise through the engine supercharger (reference 7) and is generally considered sufficiently accurate for cooling-correlation work.

By use of the results of the present tests, the change in mean effective gas temperature of the cylinder head due to a change in t_m from 0° F can be expressed as

$$\Delta T_{g_o} = 0.56\ t_m \tag{7}$$

and for the barrel

$$\Delta T_{g_o} = 0.24\ t_m \tag{8}$$

where ΔT_{g_o} is the increment in mean effective gas temperature resulting from a change in the effective dry manifold temperature from 0° F to t_m. When equations (6) and (7) are combined and the impeller tip speed is expressed in terms of the engine speed, the impeller radius, and the impeller-gear ratio, the following relation for the increment in the head mean effective gas temperature is obtained:

$$\Delta T_{g_o} = 0.56 \left[t_c + 0.01266 \left(\frac{GrN}{1000}\right)^2\right] \tag{9}$$

and for the barrel

$$\Delta T_{g_o} = 0.24 \left[t_c + 0.01266\left(\frac{GrN}{1000}\right)^2\right] \tag{10}$$

where

G impeller-gear ratio

r impeller radius, inches

N engine speed, rpm

In order to facilitate the use of equations (9) and (10) the curves of figure 4 were corrected to a carburetor dry-air temperature of 0° F by means of the head and barrel corrections found in the present tests. These corrected curves are presented in figure 8. A graphical solution for ΔT_{g_o} developed from equation (9) for the cylinder head is given in figure 9. If the impeller radius and the impeller-gear ratio are known, the value of ΔT_{g_o} can be found for any operating combination of engine speed and carburetor inlet-air temperature.

The use of the curves is illustrated by the following example, in which ΔT_{g_o} was determined when $Gr = 41.8$ inches (value at low blower for the full-scale engine), $N = 2000$ rpm, and $t_c = 100°$ F. The intersection of Gr and N is located at A. The intersection of the line parallel to reference line GH with t_c is at B. The horizontal line at B locates C; ΔT_{g_o} is read as 105° F. The value of ΔT_{g_o} obtained from figure 9 is added to the value of T_{g_d} obtained from figure 8 to give the value of T_g.

The change in head temperature resulting from a change in the head mean effective gas temperature can be evaluated from the general cooling-correlation equation as

$$\Delta T_x = \Delta T_g \left[\frac{1}{1 + (T_g - T_x)/(T_x - T_a)}\right]$$

The solutions of the foregoing equation are shown graphically in figure 10 for various values of $T_x - T_a/T_g - T_x$. Figure 10 is applicable to any temperature that may be used to represent the head temperature.

SUMMARY OF RESULTS

The results of tests to determine the effect of carburetor dry-air temperature on the cooling characteristics of a typical air-cooled engine cylinder indicate that the variation of mean effective gas temperature with carburetor dry-air temperature is independent of fuel-air ratio over the range investigated. The mean effective gas temperature varied about 1/2° F for the cylinder head and about 1/4° F for the cylinder barrel with each 1° F variation in carburetor dry-air temperature.

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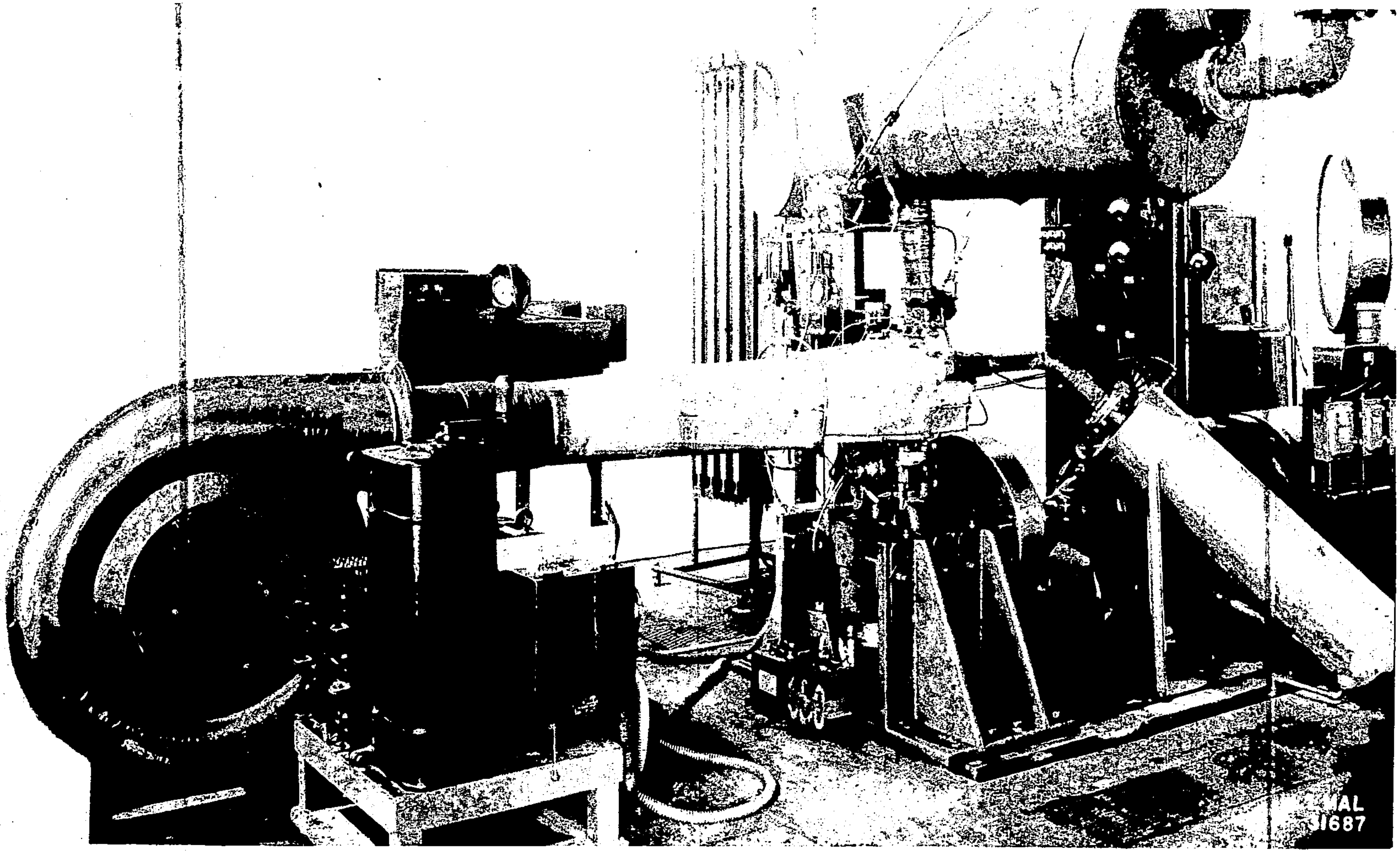


Figure 1. - Setup of single-cylinder test unit.

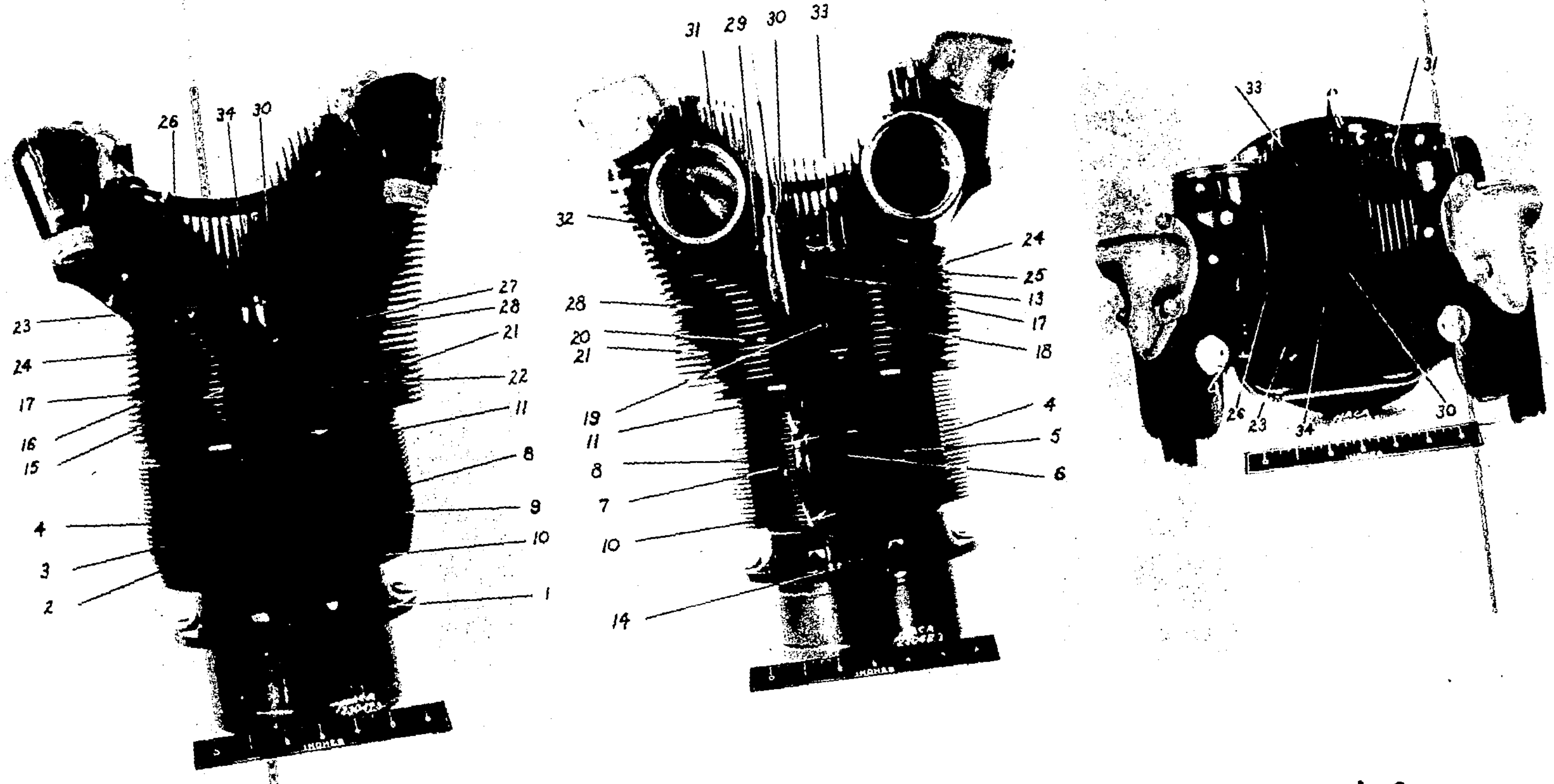
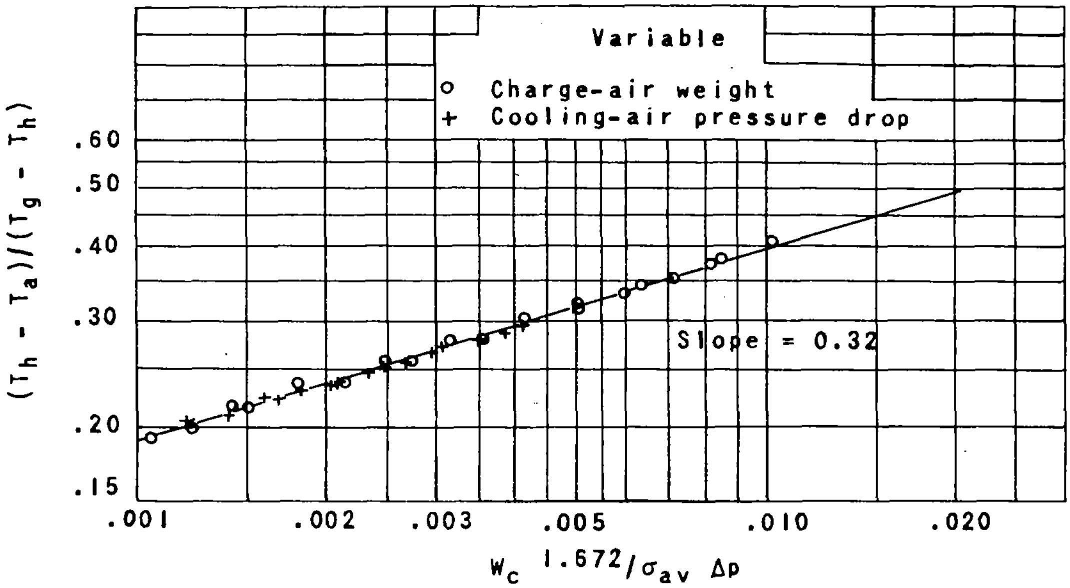
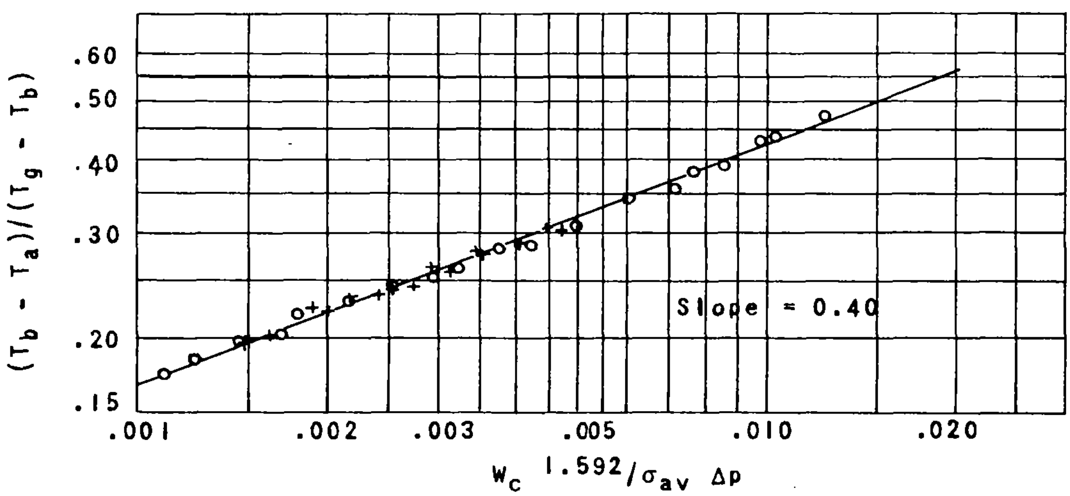


Figure 2. - Three views of test cylinder showing location of thermocouples.



(a) Cylinder head.



(b) Cylinder barrel.

Figure 3. - Cooling correlation for test cylinder.

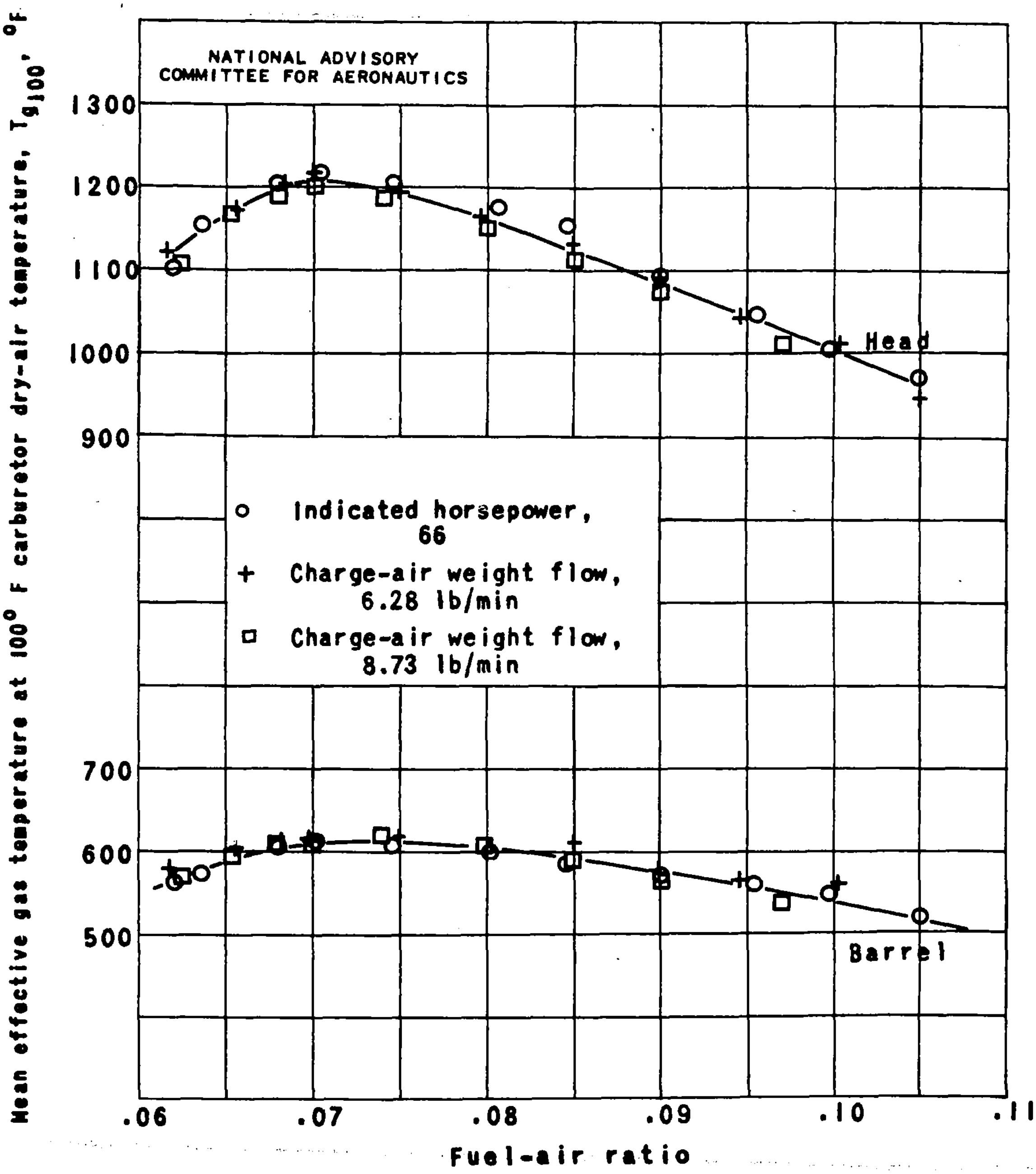


Figure 4. - Variation of mean effective gas temperature with fuel-air ratio at a carburetor dry-air temperature of 100° F.

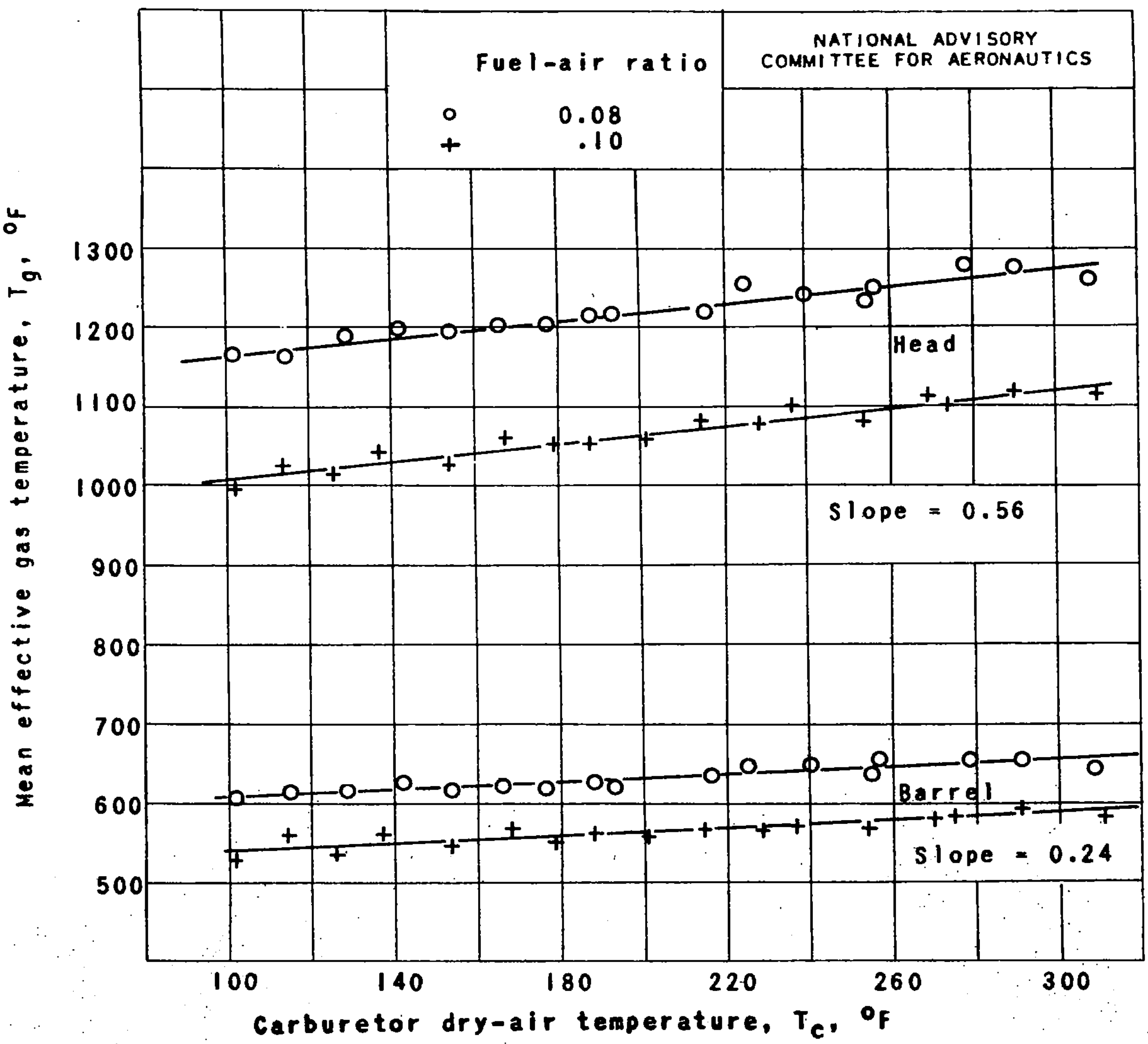


Figure 5. - Variation of mean effective gas temperature with carburetor dry-air temperature; cooling equations based on average head and average barrel temperatures.

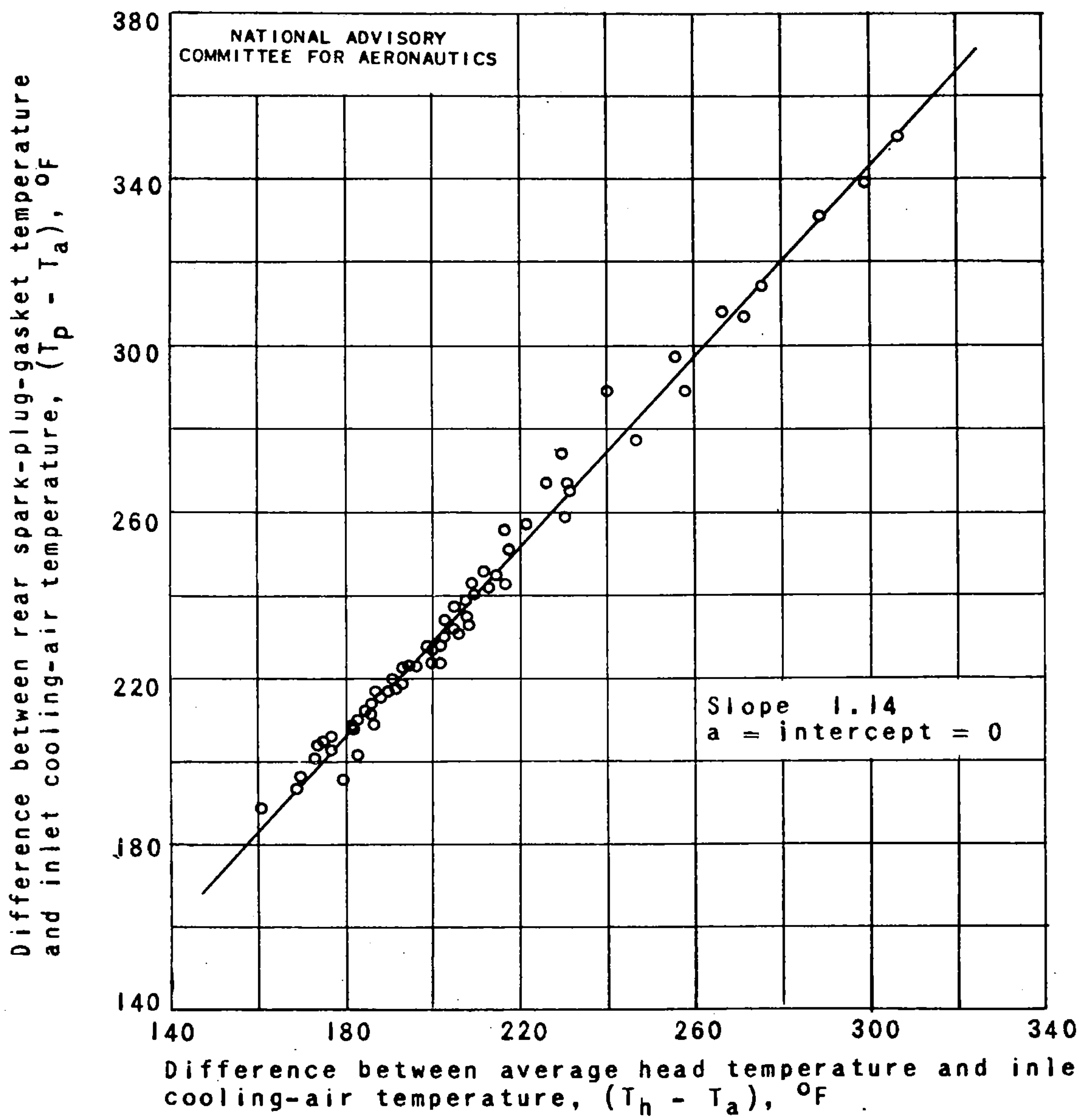


Figure 6. - Variation of rear spark-plug-gasket temperature with average head temperature.

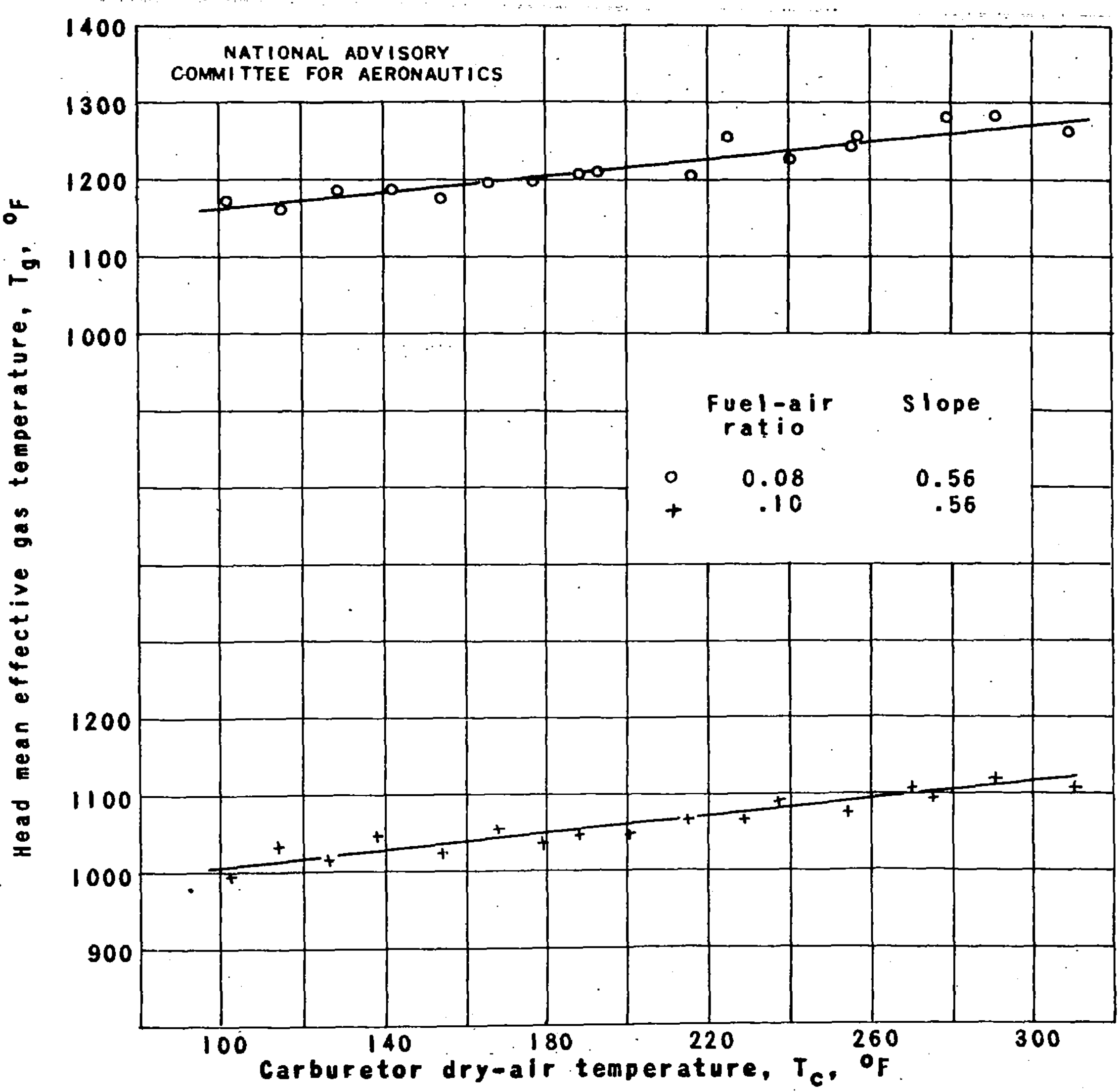


Figure 7. - Variation of head mean effective gas temperature with carburetor dry-air temperature; cooling equation based on rear spark-plug-gasket temperature.

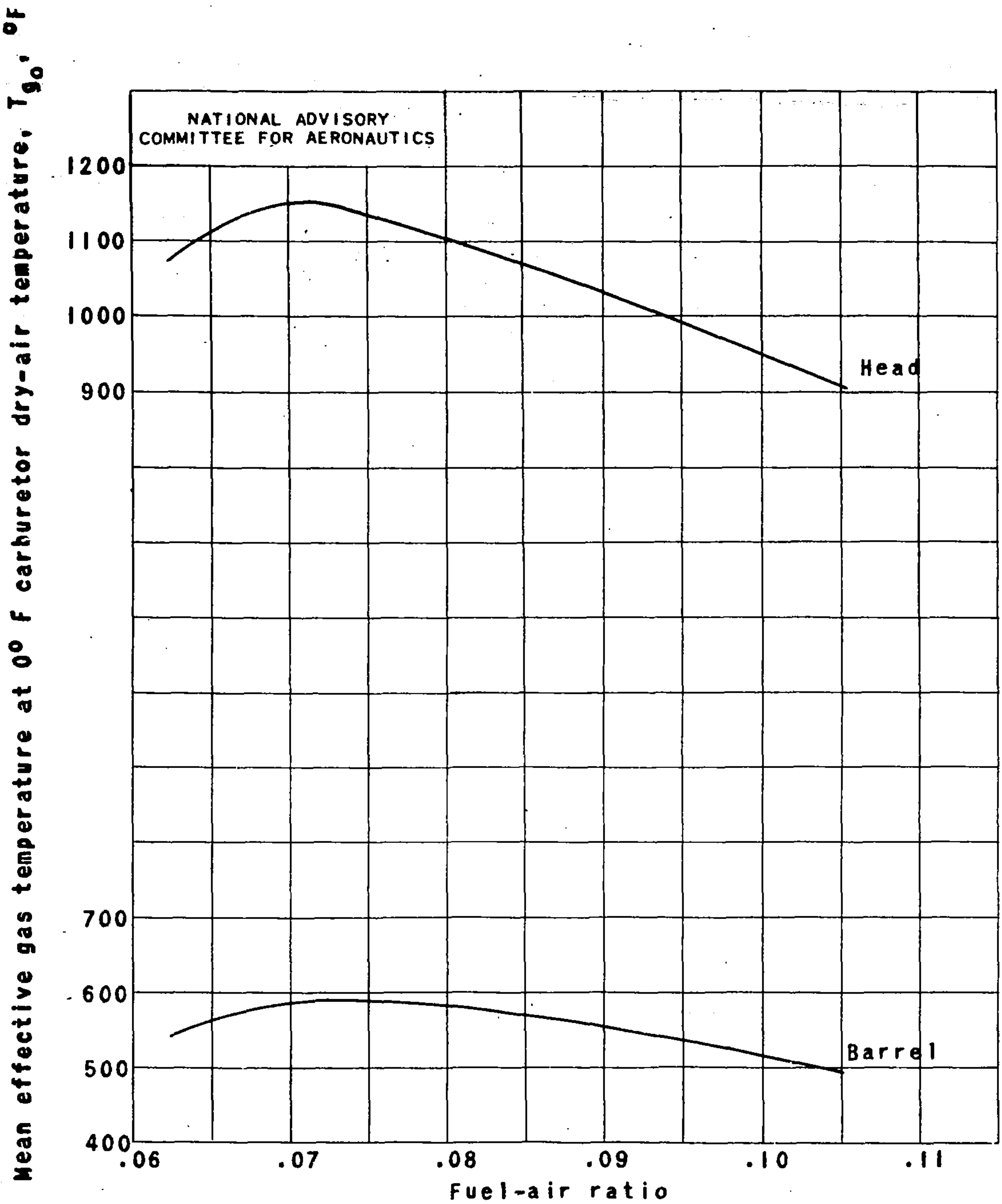


Figure 8. - Variation of mean effective gas temperature at carburetor dry-air temperature of 0° F with fuel-air ratio.

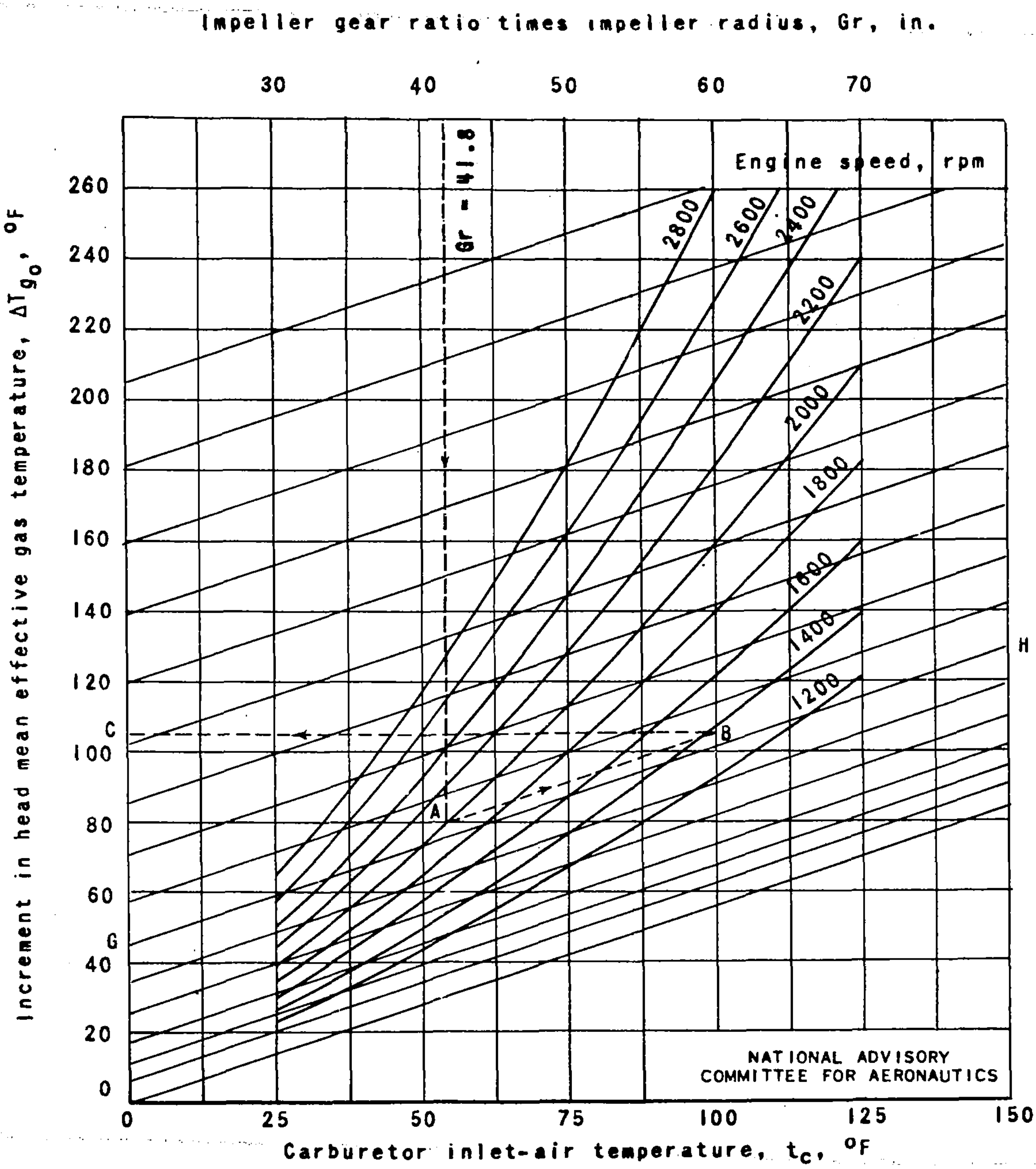


Figure 9. - Correction curves for head mean effective gas temperature.

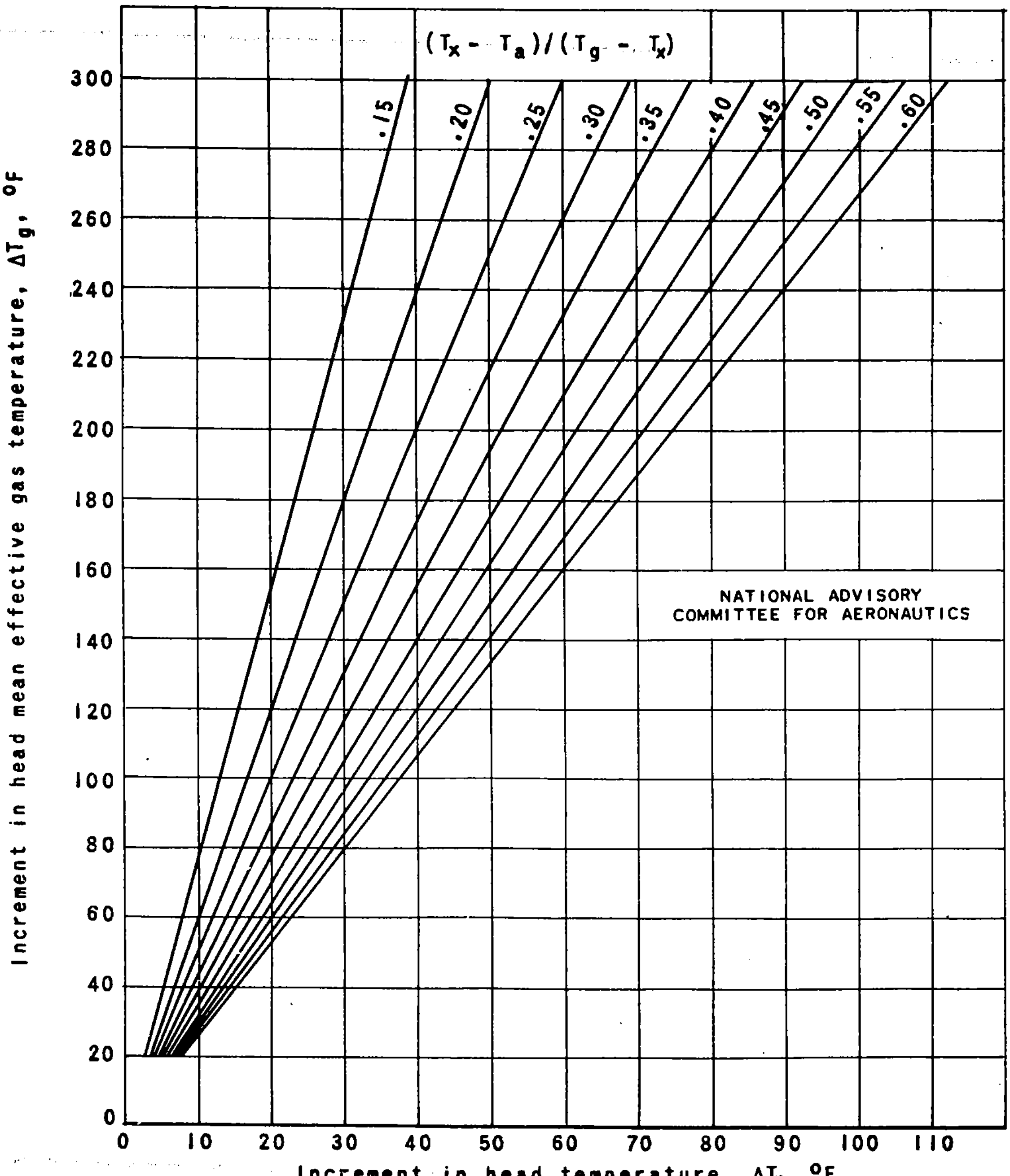


Figure 10. - Increment in head temperature resulting from an increment in head mean effective gas temperature.